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Enhancement of Efficiency of Electro-Optical Systems by Means of Optimization of Spring-Loading of EOS Actuators

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Abstract

To eliminate pitch-play in the interacting teeth of gears of EOS actuators, the gears are spring-loaded. This paper shows that traditional EOS actuators have, by an order of magnitude, a greater wear-and-tear and weight of the actuators, overheating, etc. The approach suggested hereby enables optimization of gear spring-loading and considerably improve the efficiency of EOS.

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Keywords: gears; gear system; spring-loading; EOS; resistance to wear; overheating; optimization

1. Introduction

EOS, which are used for enhancement of efficiency of radiation monitoring, detection of leaks in oil and gas pipelines, etc., are usually mounted on vessels, aerial vessels or vehicles [1]. It is known that root mean square error (RMSE) of stabilization values of the line of sight (LS) in high-precision EOS must not be greater than [2]

$$\sigma_{st} \sim 10^1 \text{ sec}. \tag{1}$$

The main design principles of high-precision EOS were formed in 60th–80th of the previous century [2–4]. Particularly, it was considered that RMSE of accuracy of manufacturing of toothed gears is limited [4]

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$$\sigma_{tooth} \sim 10^2 \sec. \quad (2)$$

First, these pitch-plays originated from the finite accuracy of tooth manufacturing, secondly, they were introduced deliberately to avoid binding of teeth of the gear assembly because of the change of the ambient temperature, temperature change due to intense operation of the EOS, etc.

It is also considered a tradition to use single-pair reduction gear systems for limitation of the dimensions of the platform actuators (PA) at the moment of zero-motion torque [2]

$$K_{red} \sim 10^{2,5}. \quad (3)$$

If there is no spring-loading in the PA, then the stabilization error because of high reduction coefficient (3) and big values of loose run with RMSE (2), calculated for driven gear respectively, may reach high values [5]

$$\sigma_{st} \sim 10^{2,5} \sec. \quad (4)$$

Compared to RMSE (1) this error (4) is intolerably large. This is the reason because of which they use spring loading of gears of the PA gear systems [6], and which inevitably results in quicker wear of EOS PA. Furthermore, it is possible that at high environmental temperatures the EOS will overheat.

In connection with it let us try to analyze interrelation of wear-resistance and spring-loading forces in the gear systems of EOS PA.

2. Estimation of influence of spring-loading on wear-resistance of EOS PA

It has been known that wear-resistance of the mechanical part C is defined as joint force F_j, which is perpendicular to the rubbing surfaces, multiplied by path length L to wearout condition [7]

$$C = F_j^{\,m} \cdot L, \quad (5)$$

where: $m = 1$ with no lubricant and $m = 3$ with lubricant,

$$F_j = F_{\text{int.}} + F_{spr.}, \quad (6)$$

$F_{\text{int.}}$ – force of interaction of the two teeth during transmission of motion, $F_{spr.}$ – spring-loading force. In accordance with c formulae (5) and (6), with lubricant ($m = 3$) and equal wearout, which depends on the materials of the rubbing surfaces, the introduction of the spring-loading reduces the path length to wearout condition (5) of the rubbing surfaces from value L_1 to value L_2

$$L_2 = L_1 \Big/ (1+s)^3, \quad (7)$$

where

$$s = F_{spr.} \Big/ F_{\text{int.}}. \quad (8)$$

In addition to the above, the optimal spring-loading force is usually chosen according to the formula [8]

$$s_{opt.} \approx 2\,. \tag{9}$$

As it follows from experimental results, wear of the spring-loaded PA may be quicker by 3 – 4 orders of magnitude, than that of non-spring-loaded ones. In accordance with formulae (7) and (9), such severe wear of the spring-loaded PA may be a result of action of spring-loading forces, coefficient of spring-loading (8) of which may be more than one half of an order of magnitude greater than the optimal values (9). It is required to optimize the spring-loading forces to the values which correspond with spring-loading coefficient (9).

Wear gain μ because of spring-loading coefficient optimization s (8) we shall define through fraction, in which: the numerator is – path length $L_{opt.}$ with applied optimized spring-loading coefficient (9), the denominator is path length $L_{cur.}$ with original spring-loading force: $\mu = L_{opt.}/L_{cur.}$. Taking into account the formula (7), this fraction may be set out as follows:

$$\mu = \left(\frac{1+s_{cur.}}{1+s_{opt.}}\right)^3 \tag{10}$$

Value s (10) may be expressed in terms of efficiency coefficient of the gear system. EOS gear system usually consists of four stages. Then, taking into account [8]

$$\eta_{gear.}(s) \approx \left(\frac{\eta_0}{1+2s\cdot(1-\eta_0)}\right)^4, \tag{11}$$

where η_0 – efficiency coefficient of one stage of the gear system without spring-loading, usually $\eta_0 \approx 0{,}8$.

For calculation of the efficiency coefficient of the gear system $\eta_{gearsyst}(s)$ (11) it is convenient to use the following formula [8]:

$$\eta_{EOS}(s) \approx \eta_{ampl}\cdot\eta_{gear.}(s)\cdot\eta_{mot.}, \tag{12}$$

where: $\eta_{ampl}, \eta_{mot.}$ – efficiency coefficients of the amplifier and motor respectively. Usually they are [2]:

$$\eta_{ampl}\cdot\eta_{mot.} \approx 1\,. \tag{13}$$

Taking into account (13), (14)

$$\eta_{gear.}(s) \approx \eta_{EOS}(s), \tag{14}$$

thereby

$$\eta_{EOS}(s) = \frac{P_{turn}}{P_{cons.}(s)} \tag{15}$$

Taking into account formulae (11), (14), (15), the spring-loading coefficient is expressible as

$$s \approx \frac{\eta_0\cdot\sqrt[4]{P_{cons.}/P_{turn}}-1}{2\cdot(1-\eta_0)} \tag{16}$$

Usually the maximal values of power are either known or may be calculated. For example, maximal power consumption value according to [2, 9-14] is

$$P_{\max.cons.} \approx \left(10^{2,5} - 10^{3}\right) W \qquad (17)$$

$P_{turn\,max.}$ here is maximal power necessary for turning of the EOS under the conditions of maximally dynamic input disturbances, caused by dynamics of the motion of the target and oscillation of the EOS base.

$$P_{turn.\max.} = P_{dyn.\max} + P_{osc.\max} \qquad (18)$$

Maximal power value $P_{turn.\max.}$ corresponds with maximal force $F_{turn.\max.}$, exerted in interaction of the teeth of the gear system of the EOS actuator [15]. As this takes place, the force of teeth interaction $F_{\mathrm{int.max}}$ is approximately equal to the force, applied to the teeth of the gear system of the PA for turning EOS, taking into account the efficiency coefficient of the corresponding stage of the gear system

$$F_{\mathrm{int.max}} \approx \frac{F_{turn.}}{\eta_{gear.}\left(s_{opt.}\right)}. \qquad (19)$$

In connection with the above-mentioned, the coefficient of spring-loading (8) at maximal force of the interaction takes minimal values

$$s_{\min} = \frac{F_{spr.}}{F_{\mathrm{int.max}}} \qquad (20)$$

Taking into account formulae (16), (19), (20), the minimal coefficient of spring-loading may be set out through maximal developed powers

$$s_{\min} \approx \frac{\eta_0 \cdot \sqrt[4]{P_{cons.\max.} / P_{turn.\max}} - 1}{2 \cdot \left(1 - \eta_0\right)} \qquad (21)$$

For the estimation of $s_{мин}$ (22) it is necessary to assess values of components $P_{turn.\max}$ (18). Let us assess the first component $P_{dyn.\max}$ (18).
Usually during steady automatic tracking of the target the velocity of the movement of the line of sight of the EOS is $\left|\alpha'_{LS}\right| < 1\, rad/s$ and acceleration rate of movement of the LS is $\left|\alpha''_{LS}\right| < 1,5\, rad/s^2$ [5, 9–20]. At these basic parameters and positioning of the data-processing units (DPU) of the EOS at the maximal distance from the EOS centre, the power necessary for turning of the EOS will not exceed the values [16, 17] $P_{dyn.\max} < 1W$, $P_{dyn.\max} >> P_{osc.\max}$, $P_{turn.\max} < 1W$.

By substituting values from (17), (18) to (21), will result in

$$s_{tek.} \approx \left(6 - 9\right). \qquad (22)$$

Advantage in optimization of the spring-loading forces by means of reduction of spring-loading coefficient to optimal values taking into account formulae (10) and (22) would allow to get advantage in wear-resistance greater than an order of magnitude

$$\mu \approx \left(10^{1}...10^{1,5}\right). \tag{23}$$

Whereby the consumed power will be reduced approximately to values

$$P_{cons.\max} \sim 2\cdot 10^{1} W. \tag{24}$$

In [16, 17] it was demonstrated, that at consumed power (24) the EOSs of radii R_{EOS} do not overheat even at high ambient temperatures $T \approx 40^{0} C$.

Furthermore it is known that usually in EOS at consumed power $P_{consumed.\max} \approx \left(10^{2,5}...10^{3}\right)W$ (17) payload – DPUs (televisual, thermal-imaging, range-measurement channels, etc.) is 4 and more times smaller than EOS actuator weight [2, 9–14],

$$m_{PA} > \frac{3}{4}\cdot m_{EOS} \tag{25}$$

Taking into account the values of standard example [15,16]

$$m_{EOS} \sim 10^{0,5} kg, \tag{26}$$

$$m_{PA} > \frac{3}{4}\cdot 10^{0,5} kg \tag{27}$$

where the mass of the DPU is

$$m_{DPU} < 10^{1} kg. \tag{28}$$

The fact that the mass of the PA (27) is many times more than the mass of the DPU (28), is because of use of excessive spring-loading force in the EOS, which leads to big spring-loading coefficients (22), and consequently to high power of the PA, which, in turn, leads to large size of the PA. Also the opposite is true: it is known that PA mass may be reduced in proportion to reduction of the power of the PA [2]. In the analyzed example, taking into account formulae (17), (24), (25), the mass of the actuators may be reduced to the following values

$$m_{PA} \approx \frac{20}{300}\cdot\frac{3}{4}\cdot m_{EOS} \tag{29}$$

If we insert value (23) into the formula (29), we get

$$m_{PA} \approx 1,5 kg \tag{30}$$

This approach allows to increase EOS PA wear out period minimally by an order of magnitude (23), to decrease mass (30) and consumed power (24) of EOS PA minimally by an order of magnitude, and eliminate overheating even at high operating temperatures.

3. Conclusions

In the interacting teeth of the gear systems of PAs of the EOS usually there are pitch-plays. First, these pitch-plays originate from the finite accuracy of tooth manufacturing, secondly, they were introduced deliberately to avoid binding of teeth of the gear assembly because of the change of the ambient temperature, temperature change due to intense operation of the EOS, etc. This leads to great EOS LS stabilization errors. For reduction of such stabilization errors the gears of the gear systems are spring-loaded. At the same time it is known that spring-loading of mechanical gears causes quicker wear of the EOS PA.

The gist of the suggested method of selection of the spring-loading coefficient is that the spring-loading coefficient is calculated for maximal input disturbances, and maximal input disturbances are defined not heuristically, but analytically.

This approach allows to increase EOS PA wear out period minimally by an order of magnitude, to decrease mass and dimensions of EOS PA minimally by an order of magnitude, and eliminate overheating even at high operating temperatures.

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